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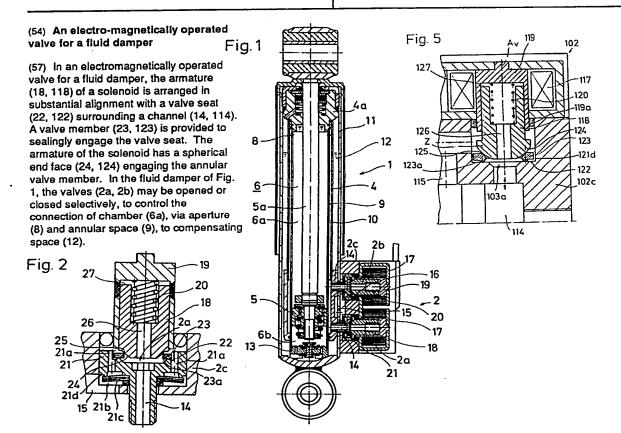
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(54) **An electro-magnetically operated
valve for a fluid damper**

(57) In an electromagnetically operated
valve for a fluid damper, the armature
(18, 118) of a solenoid is arranged in
substantial alignment with a valve seat
(22, 122) surrounding a channel (14, 114).
A valve member (23, 123) is provided to
sealingly engage the valve seat. The
armature of the solenoid has a spherical
end face (24, 124) engaging the annular
valve member. In the fluid damper of Fig.
1, the valves (2a, 2b) may be opened or
closed selectively, to control the
connection of chamber (6a), via aperture
(8) and annular space (9), to compensating
space (12).

Fig. 1

Fig. 5

Fig. 2

At least one drawing originally filed was informal and the print reproduced here is taken from a later filed formal copy.

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A FLUID PASSAGE UNIT

This invention relates to an electromagnetically operated fluid passage unit. Such a fluid passage unit is used, for example, for hydraulic vibration dampers for varying the damping force. The fluid passage unit in such a vibration damper is provided as a bypass between two fluid chambers. In such hydraulic vibration dampers, it must be ensured particularly in view of low speeds that the fluid passage unit be absolutely tight.

In a known fluid passage unit for use as a bypass in a hydraulic vibration damper, the armature of a solenoid and the valve member form a structural unit. The sealing surface of the valve member and the guiding surface of the solenoid in which the armature is guided must be manufactured with extreme precision. More particularly, it is necessary that the axis

of the armature and the axis of the magnetic core guiding the
armature be exactly perpendicular to the surface of the
valve seat. Moreover the bore in a housing accomodating the
magnetic core of the solenoid must be precisely perpendicular
to the surface of the valve seat. In order to satisfy the
demand of absolute tightness of the valve component in
closed condition, the manufacturing costs are very high.

It is the object of the present invention to provide a mag-
netically operated fluid passage unit which is absolutely
tight in its closed condition and nevertheless can be
manufactured with reduced precision demands at reduced costs.

A fluid passage unit comprises means defining a fluid channel
system and an electromagnetically operated valve system allocated
to the fluid channel system for controlling fluid flow through
the fluid channel system in at least one flow direction. The
valve system comprises a valve seat surrounding a first
fluid channel of the fluid channel system and a valve member
group cooperating with the valve seat. The valve member group
comprises an electromagnetically operated driver member and
a valve member sealingly engageable with the valve seat. The
valve member is in universal link engagement with the
driver member.

Preferably, the driver member comprises a spherical support
face, and the valve member is in engagement with the spherical
support face. The valve member may have a substantially planar
sealing face engageable with a substantially planar counter-
sealing face of the valve seat. The terms "sealing face" and
"counter-sealing face" are used here irrespective of
the shape of the valve seat. If the valve seat is formed

of an annular edge, it is regarded also as a planar sealing
face as long as this annular edge lies in a plane.

The valve member should be free for displacement with respect
to the valve seat along the counter-sealing face of the valve
seat.

The valve member may be an annular valve member which may
have a substantially planar sealing face for engagement
with a substantially planar counter-sealing face of the valve
seat.

The annular valve member may have an annular engagement edge
engaging the spherical support face. Alternatively, the annular
valve member may have an annular spherical engagement face for
engagement with the spherical support face.

The annular valve member may be sealingly engageable with the
spherical support face.

The planar sealing face or the planar counter-sealing face
may be shaped as a sealing edge of the valve member and the
valve seat, respectively.

The annular valve member may be rigid or alternatively
thin-walled and resilient.

The driver member may be biased by spring means towards a
first terminal position and may be movable by an electro-
magnetic coil towards a second terminal position. Preferably,
the driver member is biased towards a fluid flow-inter-
rupting position by spring means and is movable by an
electromagnetic coil towards a fluid flow-permitting
position.

A most simple construction is obtained if the driver member is the armature of a solenoid.

In order to provide smooth operation with a minimum of spring forces and electromagnetic force, it is desirable that the driver member be subjected to fluid pressure within the fluid channel system such that the forces resulting from the fluid pressure are substantially balanced.

The armature may be guided within a hollow core member of the solenoid while this hollow core member is surrounded by an electromagnetic coil.

The fluid passage unit of this invention is partticularly useful for vibration dampers of motor-vehicles. Such a vibration damper comprises a cylinder member and a piston rod member. At least two fluid chambers are defined within at least one of the cylinder member and the piston rod member. These fluid chambers may be connected to the fluid passage unit so that the fluid passage unit functions as a bypass which may be selectively opened or closed.

The fluid passage unit may comprise a housing rigidly fixed to the vibration damper.

Alternatively, the fluid passage unit may be connected to the fluid chamber of the vibration damper by external conduits.

The fluid passage unit of this invention may be used with double-tube vibration dampers, mono-tube vibration dampers and hydro-pneumatic suspension systems.

The various features of the invention are discussed especially in the accompanying claims which form part of the disclosure.

The invention will now be described by way of example with reference to the accompanying drawings in which:

FIGURE 1 shows a longitudinal section through a double-tube vibration damper of which the damping force can be varied by an electromagnetically operated bypass unit;

FIGURE 2 shows a valve system of the bypass unit of Figure 1;

FIGURE 3 shows a modified embodiment of the valve system;

FIGURE 4 shows a longitudinal section through a mono-tube vibration damper with an electromagnetically operated bypass unit;

FIGURE 5 shows a valve system to be used in the embodiment of Figure 4; and

FIGURE 6 shows a fluid passage unit of this invention in connection with a hydro-pneumatic suspension system.

Figure 1 shows a double-tube vibration damper 1 comprising a cylinder tube 4 and a container tube 11. A cavity 6 is provided within the cylinder tube 4. This cavity 6 is subdivided by a piston unit 5 into two working chambers 6a, 6b. the piston unit 5 is fastened to a piston rod 5a which is sealingly guided through a sealing and guiding unit 4a at the upper end of the cylinder tube 4. An annular compensating space 12 is provided between the cylinder tube 4 and the container tube 11. The working chambers 6a, 6b are filled with a damping liquid. The compensating space 12 is partially filled with the damping liquid and partially with a gas, more particularly with air.

The piston unit 5 is provided with a piston valve system which
allows liquid flow across the piston unit 5 in response to
axial movement of the piston unit 5 in both inward and outward
direction. At the lower end of the cylinder tube 4 is provided
a bottomvalve unit 13 permitting flow of damping liquid between
the lower working chamber 6b and the compensating space 12.

The valve characteristics of the piston valves and the bottom
valves can vary widely in accordance with the demands of the
vehicle manufacturers. A typical example is as follows:

On outward movement of the piston rod 5a, a high flow resi-
stance exists in the piston 5 for the damping liquid flowing from
the working chamber 6a to the working chamber 6b. As the volume
of the piston rod 5a within the cavity 6 is reduced, liquid will
flow from the compensating space 12 into the lower working
chamber 6b. The bottom valve 13 exerts only small resistance,
if any, on the flow of damping liquid from the compensating
space 12 to the working chamber 6b.

If the piston rod 5a moves inwards of the cylinder tube 4,
the volume of the piston rod 5a within the cavity 6 is in-
creased. Therefore, damping liquid must escape from the work-
ing chamber 6b towards the compensating space 12. The bottom
valve unit 13 exerts high resistance on this flow of liquid.
Simultaneously, a flow of damping liquid occurs from the lower
working chamber 6b to the upper working chamber 6a. The valve
system in the piston unit 5 exerts resistance on this flow
of liquid so that the pressure within the working chamber 6a
becomes smaller than the pressure within the working chamber 6b.
It is to be noted, however, that under normal circumstances
the pressure within the upper working chamber 6a still is
considerably higher than the pressure within the compensating
space 12.

The preceding operation of the vibration damper corresponds to a type of vibration damper the characteristics of which cannot be modified during operation.

The vibration damper of Fig. 1, however, allows such modification of the damping characteristcs, due to the provision of a bypass unit which is designated 2. The bypass unit 2 comprises two valve systems 2a and 2b. An intermediate tube 10 surrounds the cylinder tube 4; an annular space 9 is defined between the cylinder tube 4 and the intermediate tube 10. A fluid path extends from the upper working chamber 6a through a bore 8, the annular space 9, and the bypass unit 2 to the compensating space 12. This fluid path can be closed if both valve systems 2a and 2b are closed. In this case the vibration damper acts as described above. If the damping force is to be reduced more or less, the bypass unit 2 will be activated. The valve systems 2a and 2b are arranged in parallel within the fluid path extending from the upper working chamber 6a through the bore 8, the annular space 9, and the bypass unit 2 to the compensating space 12. The parallel flow branches through the valve systems 2a, 2b have different flow resistances. Thus the characteristics of the vibration damper can be varied by opening either the valve system 2a or the valve system 2b or both of them. Consequently, the vibration damper of Fig. 1 has four different damping characteristics (including the damping characteristic provided by closing both valve systems 2a and 2b).

If one of the valve systems 2a and 2b or both are opened, the operation of the vibration damper will be as follows:

On outward movement of the piston rod 5a, there is again a flow of damping liquid from the upper working chamber 6a to the lower working chamber 6b across the piston unit 5. In addition, however, there is also a flow of damping liquid from the upper working chamber 6a through the bore 8, the annular space

9, and the bypass unit 2 to the compensating space 12. As a
result, there is an increased flow of damping liquid from the
compensating space 12 through the bottom valve unit 13 into
the lower working chamber 6b. The damping force thus is
reduced as compared with the status before opening one of the
valve systems 2a, 2b.

On inward movement of the piston rod 5a, again there is a
flow of damping liquid from the lower working chamber 6b across
the piston unit 5 to the upper working chamber 6a and a flow
of damping liquid from the lower working chamber 6b to the
compensating space 12.

However, as the pressure within the upper working chamber 6a
is higher than the pressure within the compensating space 12,
there is additional flow of damping liquid from the upper
working chamber 6a through the bore 8, the annular space 9,
and the bypass unit 2 to the compensating space 12. As a
result, the damping force is reduced also on inward movement
of the piston rod 5a.

The bypass unit 2 comprises a housing 2c which is fastened
to the container tube 11, for example by welding. In the
housing 2c, a fluid channel system is provided. This fluid
channel system comprises channels 14 extending from the annular
space 9 to the respective valve systems 2a, 2b,   and a channel 15 extend-
ing from the valve systems 2a, 2b through a bore 16 to the
compensating space 12. Each of the valve systems 2a, 2b
comprises an electromagnetic coil 17.

In Fig. 2 the valve system 2a is illustrated in detail by
omitting the respective coil 17.

The valve system 2a comprises a seat member 21 with a valve seat 22 and the channel 14 provided in the center of the valve seat 22. The seat member 21 is accomodated within the housing 2c. An annular valve member 23 of substantially rectangular cross-section engages at the lower face 23a thereof the valve seat 22. An edge 25 of the valve member 23 is in engagement with a spherical surface 24 of an armature 18. This armature 18 is guided within a core 19. A spring 20 is supported by the core 19 and acts onto the armature 18 so as to press the armature 18 against the valve member 23, and the valve member 23 against the valve seat 22.

In Fig. 2, the valve system 2a is closed by the action of the spring 20. The valve member 23 is in sealing engagement with the valve seat 22 and the spherical surface 25.

When the respective electromagnetic coil 17 is energized, the armature 18 is moved upwards against the action of the spring 20. Then the valve system 2a is opened by the spherical face 24 lifting from the edge 25 of the valve member 23. The damping liquid can flow from the channel 14 across the valve system 2a to the channel 15.

It can be seen from Fig. 2 that the liquid flowing from channel 14 to channel 15 passes a group of bores 21a and a damping resistance 21b provided by a group of resilient discs 21c which are in resilient contact with an annular edge 21d. The liquid flowing from channel 14 to channel 15 lifts the resilient discs 21c from the edge 21d. The biasing force biasing the resilient dics 21c against the edge 21d determines the flow resistance of the flow path across the valve system 2a.

Fig. 2 shows a central bore 26 extending through the armature 18 so that the upper face 27 of the armature 18 is subjected to the pressure within the channel 14. Thus the forces resulting from the pressure in channel 14 and acting on the armature 18 are substantially balanced.

The valve system 2b of Fig. 1 is identical with the valve system 2a as shown in Fig. 2 and described above. The only difference possibly resides in the biasing force of the resilient discs 21 acting on the edge 21d so that the flow resistances of the valve systems 2a, 2b are different.

Fig. 3 shows an alternative embodiment of the valve system 2. Analogous components are designated by the same reference numerals as in Fig. 2. Only the shape of the valve member 28 has been modified as compared with the valve member 23 in Fig. 2. The valve member 28 is a thin-walled elastic valve member which resiliently absorbs the closure movement of the armature 18 so that closure noises are eliminated. The valve member 28 is resiliently deformable to a screen-shape so that it bears accurately on the inner diameter edge of the valve seat 22 to the effect that the surface subjected to pressure in channel 14 is accurately defined.

It can be seen from both Fig. 2 and Fig. 3 that valve member 23 and valve member 28, respectively, can tilt with respect to the spherical face 24. Moreover a radial displacement of the valve members 23 and 28, respectively, is possible relative to the seat member 21. Thus absolute tightness is warranted even if the axis of the armature 18 is not precisely aligned with the axis of the valve seat 22.

Fig. 4 shows a mono-tube vibration damper 29 comprisng a cylinder tube 29a, a piston rod 5a, and a piston 30. The

cavity 31 within the cylinder tube 29a is subdivided by
the piston 30 into an upper working chamber 31a and a lower
working chamber 31b. The lower working chamber 31b is
adjacent a compensating chamber 32 which contains pressurized
gas and is separated from the lower working chamber
31b   by   a floating separating piston 32a. This is the
conventional type of a mono-tube vibration damper. Inward
and outward movement of the piston rod 5a is damped by damping
liquid flowing through piston 30. The damping resistance for
both directions of movement may be different.

The cylinder tube 29a is surrounded by an outer tube 29b. An
annular space 29c exists between. the cylinder tube 29a and the
outer tube 29b. This annular space 29c is subdivided by a
separating ring 29d into an upper space section 29c1 and a
lower space section 29c2. The upper space section 29c1 communi-
cates through a bore 34 with the upper working chamber 31a
whereas the lower space section 29c2 communicates with the lower
working chamber 31b through a bore 33. A bypass unit 2 is
fastened to the outer tube 29b. This bypass unit 2 includes a
valve system 2a. One side of this valve system is connected
to the upper space section 29c1 through a bore 35 while the
other side of the valve system 2a is connected to the lower
space section 29c2 through a channel 14. The valve system 2a
is electromagnetically operated by an armature 18 acting on a
valve member 36.

The basic construction of the armature 18 and the valve member 36
correspond to the construction shown in Fig. 2. More parti-
cularly, there is provided again a spherical face on the
armature 18 cooperating with the annular valve member 36.

A preferred embodiment of a bypass unit for a mono-tube
vibration damper is illustrated in Fig. 5.

In Fig. 5, analogous parts are designated by the
same reference numerals as in Fig. 2 increased by 100.

Fig. 5 again shows the housing 102c of the bypass unit 102 in
which housing are provided a channel 114 and a channel 115.
Moreover the housing accomodates a magnetic core 119 sur-
rounded by the electromagnetic coil 117. In the electromagnetic core
119 is guided the armature 118 having the spherical face
124. The spherical face 124 is engageable with the annular
valve member 123, the edge 125 of which engages the spherical
face 124. The annular valve member 123 is provided with a
sealing edge 123a which engages a planar sealing face 122 of
the housing 102c. A helical compression spring 120 acts onto
the armature 118 and is supported by the core 119. The helical
compression spring is pretensioned when the valve 103a is
closed. The annular valve member 123 is a rigid valve member
which may be fabricated by stamping, coining, swaging or
sintering. Again the annular valve member 123 has radial play
with respect to the bore 121d of the housing 102c.

Again there is a channel 126 which allows the damping liquid from
channel 114 to act on the end face 127 of the armature 118.

It is to be noted that the edge 125 and the sealing edge
123a both are located on a virtual cylinder Z which is
coaxial with the axis A of the armature 118 to the effect
that the armature 118 and the annular valve member 123 are
substantially fully balanced with respect to the damping
liquid pressures in both the channels 114 and 115. That
is of considerable importance because in a mono-tube
vibration damper, as shown in Figs. 4 and 5, it is necessary
that no pressure-dependent forces act on the armature 118.

A sealing ring 119a is provided between the core 119 and the
housing 102c.

Fig. 6 shows a bypass unit 202 with valve systems 202a and 202b.
A vibration damper or suspension element 201 is connected through
a line L to a spring container 243. The spring container 243

is subdivided by a membrane 243a into a chamber 243b con-
taining pressurized gas and a chamber 243c containing pres-
surized damping liquid. The line L includes a damping unit
240 having a first damping disc 240a and a second damping
disc 240b. The design of both damping discs is identical so
that the description of the damping disc 240a will be
sufficient for an explanation. The damping disc 240a acco-
modates a first damping bore 240a1 and a second damping bore
240a2. Both damping bores 240a1 and 240a2 are provided with
check valve members 240a3 and 240a4, respectively, so that
the flow resistance across the damping disc 240a may be
different in response to the flow direction. It can be seen
from Fig. 6 that the damping discs 240a and 240b with
the respective damping bores are arranged in series between
the vibration damper 201 and the spring container 243.
When both valve systems 202a,202b are closed, the damping force
is solely defined by the damping bores of the damping
discs 240a and 240b. The valve system 202a corresponds
to the valve system 2a of Fig. 2. The valve system 202a
is provided in a bypass channel system 214a, 215ab which
is a bypass for the damping disc 240a. The valve system
202b is located within a channel system 215ab, 214b which
is a bypass for the damping disc 240b.

Four conditions are possible:

a) Both valve systems 202a and 202b are closed - none of the
   bypasses is open.

b) The valve system 202a is open - the damping disc 240a is
   bypassed.

c) The valve system 202b is open - the damping disc 240b
   is bypassed.

d) Both valve systems 202a and 202b are open - both damping
   discs 240a, 240b are bypassed.

Thus four different damping characteristics are possible
in this embodiment.

As is the case according to Fig. 2, the forces resulting
from the pressure in line L and acting on the armature 218
are balanced.

Specific forms of embodiments of the invention have been
represented and described in order to illustrate the use
of the principles of the invention. Of course, the invention
can be realized also in other ways without departing
from these principles.

The reference numbers in the claims only serve to facili-
tate the understanding and are not to be understood as a
limitation.

In Fig. 6, the sealing faces between the valve member 223 and
the valve seat 222 on the one hand and between the sealing
member 223 and the spherical face 224 on the other hand are
identical so that again a full compensation of the pressure
forces acting on the armature 218 and the valve member 223 is
obtained. The valve member 223 again may be fabricated by
stamping, coining, swaging or sintering.